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Oil-Flow Distribution  
Schemes on the Operating  
Characteristics of a  
35-Millimeter-Bore Ball  
Bearing to 2.5 Million DN

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Effect of Two Inner-Ring  
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35-Millimeter-Bore Ball  
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## Summary

Parametric tests were conducted in a high-speed bearing tester on a 35-mm-bore, split-inner-ring ball bearing with a double-inner-land-guided cage. Provisions were made for through-the-inner-ring lubrication. Tests were performed at a thrust load of 667 N (150 lb) and a combined load of 667 N (150 lb) thrust and 222 N (50 lb) radial. In some tests 50 percent of the oil supplied to the inner ring flowed radially through the bearing to the inner-ring raceway to lubricate the bearing, and 25 percent flowed axially through grooves in the bearing bore to cool the inner ring. The remaining 25 percent flowed radially through the bearing for cage-land lubrication. In other tests this flow distribution was 75 percent radially, none axially, and 25 percent radially for cage-land lubrication. Shaft speeds were a nominal 32 000 to 72 000 rpm. The oil-inlet temperature was 394 K (250 °F). Oil was supplied to the inner ring at flow rates of 341 to 1894 cm<sup>3</sup>/min (0.09 to 0.50 gal/min). Outer-ring cooling oil flow rates of 0 to 1932 cm<sup>3</sup>/min (0.51 gal/min) at a 394 K (250 °F) oil-inlet temperature were used in some tests.

The bearings were successfully operated to 2.5 million DN. Inner-ring temperatures were similar for bearings with either 75- or 50-percent flow to the raceway. The 25-percent axial flow used for cooling the inner ring was essentially ineffective. Outer-ring temperature was generally lower with 75- than with 50-percent flow to the inner-ring raceway both with and without outer-ring cooling. Cooling the outer ring did not appreciably affect the inner-ring temperature. However, the outer-ring absolute temperature decreased as much as 7 percent at 2.5-million-DN shaft speed, without any sacrifice in power loss or cage slip. Slightly higher bearing temperatures (maximum difference, 6 deg K (11 deg F)) resulted with a combined radial and thrust load than with thrust load only.

The maximum power loss obtained for the two lubrication schemes was 3.0 kW (4.0 hp) as determined from heat rejection to the lubricant. This maximum occurred at a thrust load of 667 N (150 lb), a shaft speed of 72 400 rpm, and a lubricant flow rate of 1856 cm<sup>3</sup>/min (0.49 gal/min) with 50-percent flow to the inner-ring raceway and outer-ring cooling.

Percent cage slip increased with increasing shaft speed and lubricant flow rate. Adding a 222-N (50-lb) radial

load to the 667-N (150-lb) thrust load increased the percent cage slip by a maximum of 1.8 percent. Percent cage slip was higher at 75- than at 50-percent lubricant flow to the inner-ring raceway, with a maximum difference of 1.6 percent. Maximum cage slip recorded for all tests was 8.7 percent. The largest drop in percent cage slip due to outer-ring cooling was 2.2 percent at 46 500 rpm, 1900 cm<sup>3</sup>/min (0.50 gal/min), 50-percent radial flow to the inner raceway, and 667 N (150 lb).

## Introduction

Mainshaft bearings in present-production large gas turbine engines perform satisfactorily to 2.3 million DN. (DN is defined as the speed of the shaft in revolutions per minute multiplied by the bearing bore in millimeters.) Small-bore bearing experiments in the past have generally concentrated on tests at DN values of 1.8 million or less. Reference 1 reports on 20-mm-bore ball bearings at DN values to 1.8 million and focuses on the proper techniques for scavenging test-bearing lubricating oil to reduce bearing operating temperature. Reference 2 deals with the friction and surface damage aspects of high-speed ball bearing operation to 1.0 million DN. Reference 3 reports the minimum oil requirements of 30- and 75-mm-bore ball bearings at DN values to 0.975 million.

Small advanced turbine engines (1- to 10-lb/sec total airflow) require bearings that operate in the 2.5-million-DN range at high temperatures. The bearing designs and lubrication techniques used for these engines must be refined and optimized for reliable performance and long life.

The performance of 35-mm-bore, angular-contact ball bearings with a single-outer-land-guided cage was investigated in parametric tests reported in reference 4. The investigation included lubrication by oil jets or through passages in the bearing inner ring. When inner-ring lubrication was used, the oil was channeled through axial grooves and radial holes in the bearing inner ring. In some tests 50 percent of the oil supplied to the inner ring flowed radially through the bearing for lubrication and 50 percent flowed axially to cool the inner ring. In other tests the distribution was 25 percent for lubrication and 75 percent for cooling.

The primary objective of this study was to determine the operating characteristics of a 35-mm-bore, split-inner-ring ball bearing under various coolant and lubricant flow rates. In some tests 75 percent of the bearing total oil flow was supplied to the raceway; in other tests 50 percent was supplied to the raceway. A secondary objective was to compare the effect of outer-ring cooling on bearing operating characteristics.

The bearing had a nominal unmounted contact angle of  $24^\circ$  and a double-inner-land-guided cage. Provisions were made for inner-ring lubrication of the bearing and for outer-ring cooling. Test conditions included a thrust load of 667 N (150 lb) and combined radial and thrust loads of 222 and 667 N (50 and 150 lb), respectively. Nominal shaft speeds were 32 000 to 72 000 rpm, with an oil-inlet temperature of 394 K (250 °F). Lubricant was fed to the bearing through the inner ring at flow rates from 341 to 1894 cm<sup>3</sup>/min (0.09 to 0.50 gal/min). Outer-ring cooling oil flow rates were 0 to 1894 cm<sup>3</sup>/min (0.50 gal/min). The lubricant was a tetraester that met the MIL-L-23699 specifications.

## Apparatus and Procedure

### High-Speed Bearing Tester

A general view of the air-turbine-driven test machine is shown in figure 1. The sectioned drawing in figure 2 shows a horizontally mounted shaft supported by two preloaded, angular-contact ball bearings. The test bearing was overhung and mounted in a separate housing

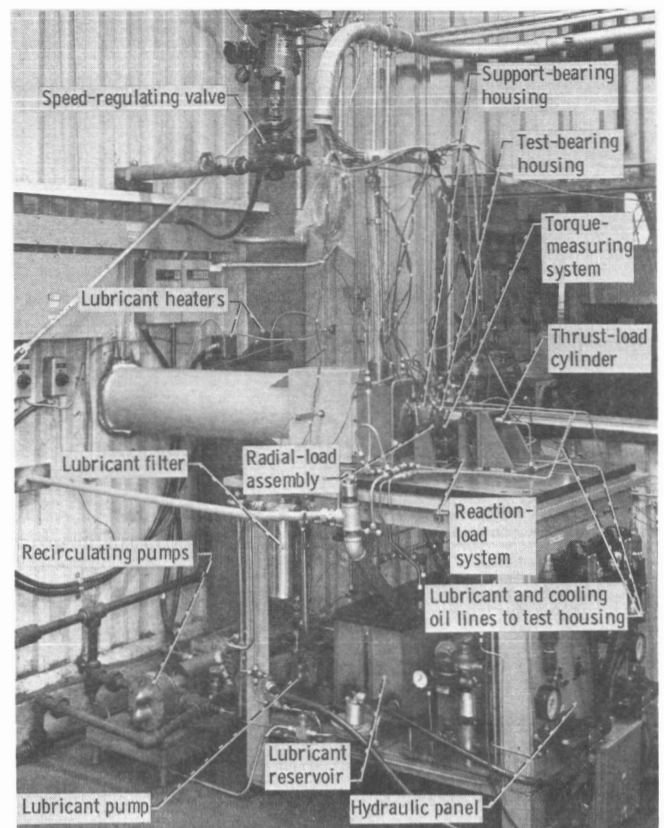


Figure 1.—High-speed, small-bore-bearing test machine.

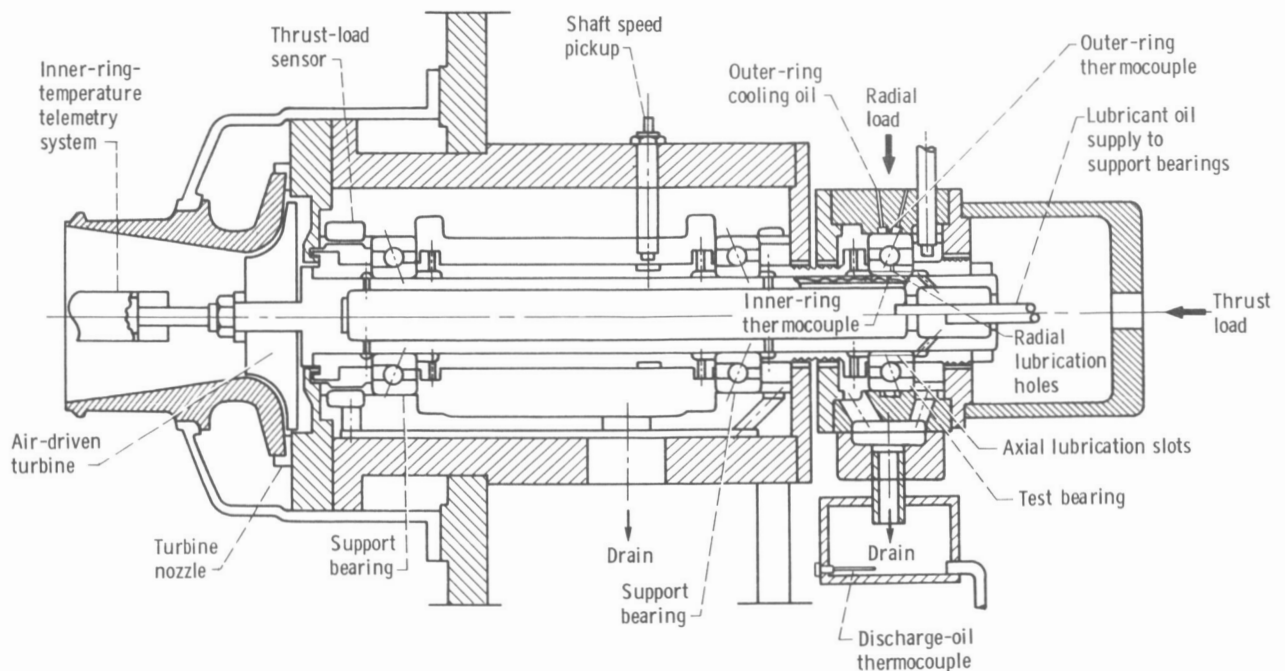


Figure 2.—Schematic of high-speed, small-bore-bearing test machine.

that incorporated the hardware for lubrication, oil removal, thrust and radial load application, and instrumentation for cage speed measurement. Test bearing torque was measured with strain gages located near the end of an arm that prevented the housing from rotating. Thrust force was applied through a combination of a thrust needle bearing and a small roller support bearing in order to minimize test-housing restraint during torque measurements. The test bearing was lubricated through the inner ring. Oil was pumped by centrifugal force from the center of the hollow shaft through axial grooves in the test bearing bore and through a series of small radial holes, 0.762 mm (0.030 in.) in diameter, to the bearing inner race. Those axial grooves in the bearing bore that did not have radial holes allowed oil to flow under the ring for inner-ring cooling. To vary the distribution of the total oil flow for lubrication and for inner-ring cooling, appropriate radial holes and axial grooves were plugged during test bearing installation. Cooling oil was supplied to the outer ring by means of holes and grooves in the bearing housing (fig. 2).

Shaft speed (inner-ring speed) was measured with a magnetic probe. Ball-pass frequency (cage speed) was determined by analyzing signals from a semiconductor strain gage mounted on the inner surface of the test-bearing housing. Two thermocouples were assembled in the shaft to measure inner-ring temperatures through a rotating telemetry system. Outer-ring temperatures were obtained by two thermocouples installed in the test bearing housing. The high-speed bearing tester is described in detail in reference 5.

### Test Bearing

The test bearing (fig. 3) was an ABEC-7 grade, 35-mm-bore ball bearing with a split inner ring and a

double-inner-land-guided cage, as shown in figure 4. The bearing contained 16 balls, each with a nominal diameter of 7.14 mm (0.281 in.). The bearing design permitted lubrication through the inner ring by means of axial grooves machined in the bore. Radial holes of 0.762-mm (0.030-in.) diameter radiating from the bearing bore formed a flowpath for bearing lubrication. The bearing inner-ring grooves and radial holes were designed so that 50 percent of the oil supplied to the inner ring lubricated the bearing, 25 percent lubricated the land surfaces, and 25 percent flowed axially through those grooves that contained no radial holes. The latter flow cooled the inner ring. This oil-flow distribution is illustrated in figure 5(a). In some tests appropriate axial grooves and radial holes were blocked to allow 75 percent of the total flow to be used for bearing lubrication, 25 percent for

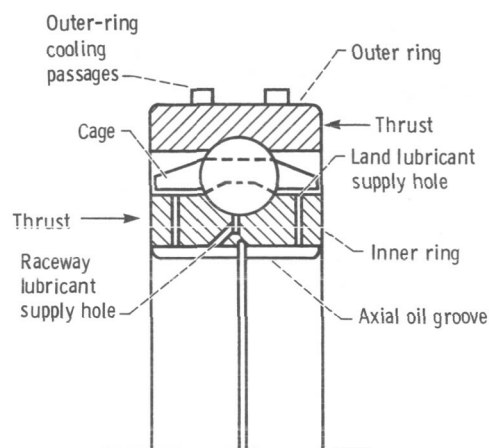


Figure 4.—Cross section of bearing showing lubricant and cooling passages.

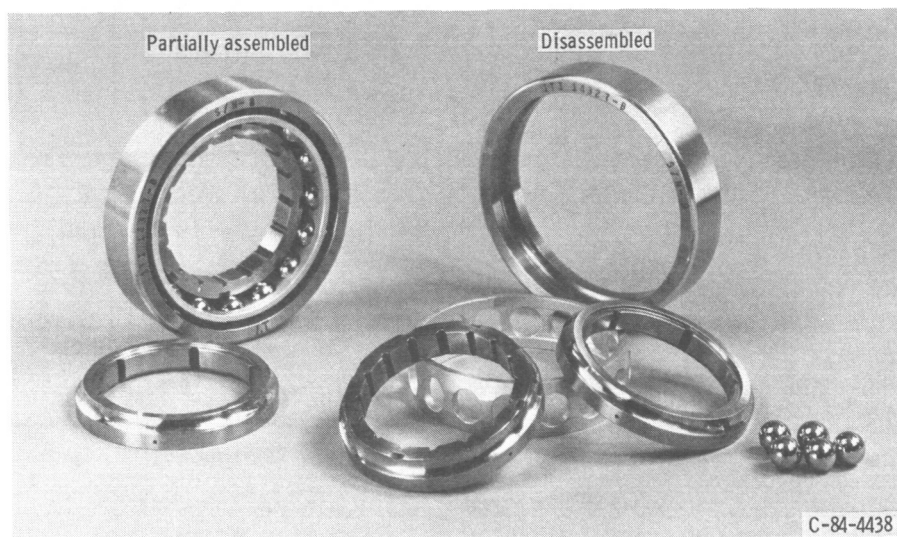


Figure 3.—35-mm-bore, deep-groove split-inner-ring ball bearing with double-inner-land-guided cage.

land surface lubrication, and none for inner-ring cooling, as shown in figure 5(b).

The inner and outer rings and the balls were manufactured from consumable-electrode-vacuum-melted AISI M-50 steel. The nominal hardness of the balls and rings at room temperature was Rockwell C62. The cage was made from AISI 4340 steel (AMS 6415) heat treated to Rockwell C28 to C36 hardness and completely plated with silver (AMS 2412) 0.0203 to 0.0381 mm (0.0008 to 0.0015 in.) thick. The cage balance was within 0.05 g-cm ( $7 \times 10^{-4}$  oz-in). More complete specifications are shown in table I.

**Lubricant**

The oil used was a tetraester type II oil qualified to MIL-L-23699 specifications. The major properties of the oil are presented in table II.

**Test Procedure**

After the test machine had been warmed by recirculating heated oil and the torque-measuring system had been calibrated, a 667-N (150-lb) thrust load and a 1894-cm<sup>3</sup>/min (0.50-gal/min) lubricant flow rate were applied. The shaft speed was then slowly brought up to a nominal 28 000 rpm. When the bearing and test machine temperatures stabilized (after 20 to 25 min), the oil-inlet temperature and lubricant flow rate were set and the speed was increased to the desired value.

A test series was run by starting at the lowest nominal speed, 32 000 rpm, and progressing through 50 000,

TABLE II.—PROPERTIES OF TETRAESTER LUBRICANT

Additives.....	Corrosion and oxidation inhibitors; and antiwear and antifoam additives
Kinematic viscosity, cS, at—	
311 K (100 °F).....	28.5
372 K (210 °F).....	5.22
477 K (400 °F).....	1.31
Flashpoint, K (°F).....	533 (500)
Autogenous ignition temperature, K (°F).....	694 (800)
Pour point, K (°F).....	214 (-75)
Volatility (6.5 hr at 477 K (400 °F)), wt %.....	3.2
Specific heat at 372 K (210 °F),	
J/kg K (Btu/lb °F).....	2140 (0.493)
Thermal conductivity at 477 K (400 °F),	
J/m sec K (Btu/hr ft °F).....	0.13 (0.075)
Specific gravity at 372 K (210 °F).....	0.931

65 000, and 72 000 rpm before changing the lubricant flow rate. At each speed and flow condition a separate test was run during which the outer-ring cooling oil flow was adjusted to achieve equal inner- and outer-ring temperatures. Four lubricant flow rates of 341 to 1894 cm<sup>3</sup>/min (0.09 to 0.50 gal/min) were used. The first series of tests was run with the lubrication distribution scheme shown in figure 5(a), in which 50 percent of the total oil flow supplied to the inner ring lubricated the bearing and 25 percent flowed axially through the bearing and cooled the inner ring.

After these test runs were completed, other tests were performed to determine the effects of

TABLE I.—TEST BEARING SPECIFICATIONS

<b>Bearing:</b>	
Bore, mm (in.).....	35 (1.3780)
Outside diameter, mm (in.).....	62 (2.4409)
Width, mm (in.).....	14 (0.5512)
<b>Cage:</b>	
Diametral land clearance, mm (in.).....	0.305 (0.012)
Diametral ball pocket clearance, mm (in.).....	0.660 (0.026)
Material.....	4340 per AMS 6415 (silver plated)
Hardness.....	Rockwell C28-36
<b>Balls:</b>	
Number.....	16
Size (diameter), mm (in.).....	7.14 (0.28)
Grade.....	10
Material.....	CEVM M-50 per AMS 6490
Hardness.....	Rockwell C60 (min.)
<b>Race:</b>	
Inner conformity, percent.....	54
Outer conformity, percent.....	52
<b>Assembly:</b>	
Internal radial clearance, mm (in.).....	0.061 (0.0024)
Contact angle, deg.....	24

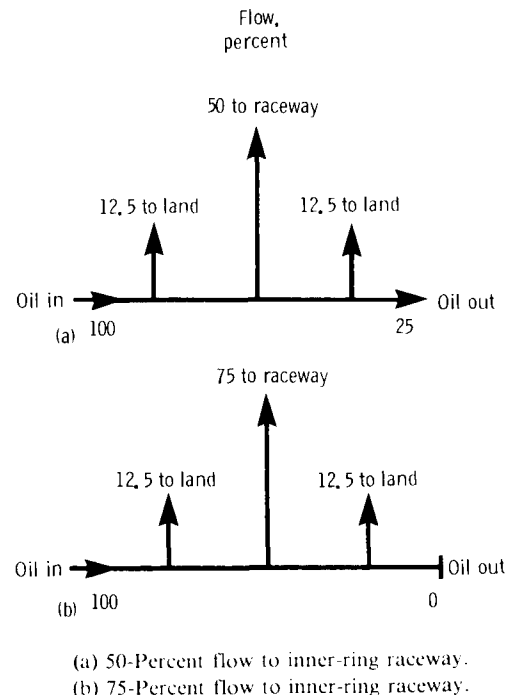


Figure 5.—Schematic of oil-flow distribution through inner ring.

(1) Adding a nominal 222-N (50-lb) radial load to the 667-N (150-lb) thrust load

(2) Changing the oil-flow distribution scheme to that shown in figure 5(b) in which 75 percent of the total flow supplied to the inner ring lubricated the bearing, with no axial oil flow through the bearing to cool the inner ring. A nominal thrust load of 667 N (150 lb) was used for this series of tests. For each of these conditions tests were run at nominal speeds of 50 000, 65 000, and 72 000 rpm.

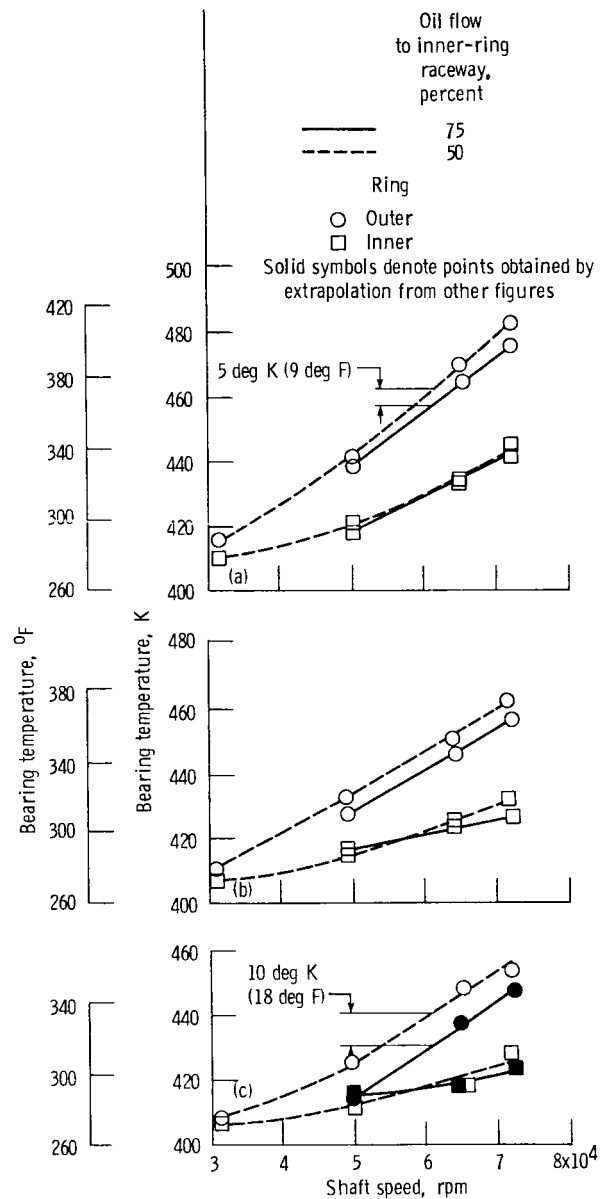
If it became apparent during the course of testing that the conditions would result in predictable distress of the test bearing or test rig or that the bearing temperature would exceed 491 K (425 °F), the test point was aborted or omitted.

## Results and Discussion

Parametric tests were conducted with a 35-mm-bore, split-inner-ring ball bearing having a double-inner-land-guided cage. Shaft speeds ranged from 32 000 to 72 000 rpm. Test conditions included an oil-inlet temperature of 394 K (250 °F) and oil flow rates from 341 to 1894 cm<sup>3</sup>/min (0.09 to 0.50 gal/min). Outer-ring cooling oil flow rates from 0 to 1932 cm<sup>3</sup>/min (0.51 gal/min) at 394 K (250 °F) oil-inlet temperature were used in some tests. Over this entire range of test conditions the bearing operated successfully without any visible damage to the components.

### Effect on Bearing Temperature of Oil-Flow Distribution Through Inner Ring

The effect on bearing temperature of the distribution of lubricant through the inner ring was determined without and with outer-ring cooling (figs. 6 and 7, respectively). Bearing temperature increased with increasing speed for all conditions investigated. The rate of increase in bearing temperature with increasing speed was greater for the bearing outer ring than for the inner ring over the range of speed and lubricant flow rates tested (fig. 6). Inner-ring temperatures were nearly the same for 75- and 50-percent flow to the raceway, indicating that the 25-percent axial flow for cooling the inner ring with 50-percent flow to the raceway was essentially ineffective. Outer-ring temperatures were lower at 75-percent flow to the raceway. The difference in outer-ring temperature  $\Delta T$  between 75- and 50-percent flow to the raceway increased with increasing lubricant flow rate over the speed range tested. At a total flow rate of 758 cm<sup>3</sup>/min (0.20 gal/min) with 75-percent flow to the inner-ring raceway (fig. 6(a)), the outer ring was on average about 5 deg K (9 deg F) cooler than with 50-percent flow to the raceway. At a total flow rate of 1894 cm<sup>3</sup>/(0.50 gal/min) (fig. 6(c)), this difference in temperature increased to 10 deg K (18 deg F).



(a) Lubricant flow rate, 758 cm<sup>3</sup>/min (0.20 gal/min).

(b) Lubricant flow rate, 1330 cm<sup>3</sup>/min (0.35 gal/min).

(c) Lubricant flow rate, 1894 cm<sup>3</sup>/min (0.50 gal/min).

Figure 6.—Effect of shaft speed on test bearing temperature for two oil-flow distribution schemes—no outer-ring cooling. Thrust load, 667 N (150 lb).

Some applications may require (for best performance and optimal operating clearance) a minimal temperature gradient between the bearing inner and outer rings. Therefore tests were conducted to find the outer-ring cooling oil flow rates that produced a thermally balanced bearing, with equal inner- and outer-ring temperatures. When the results of tests with 75- and 50-percent flow to the inner-ring raceway were compared (fig. 7), bearing temperature was generally lower with 75-percent flow to the raceway. The temperature difference between the

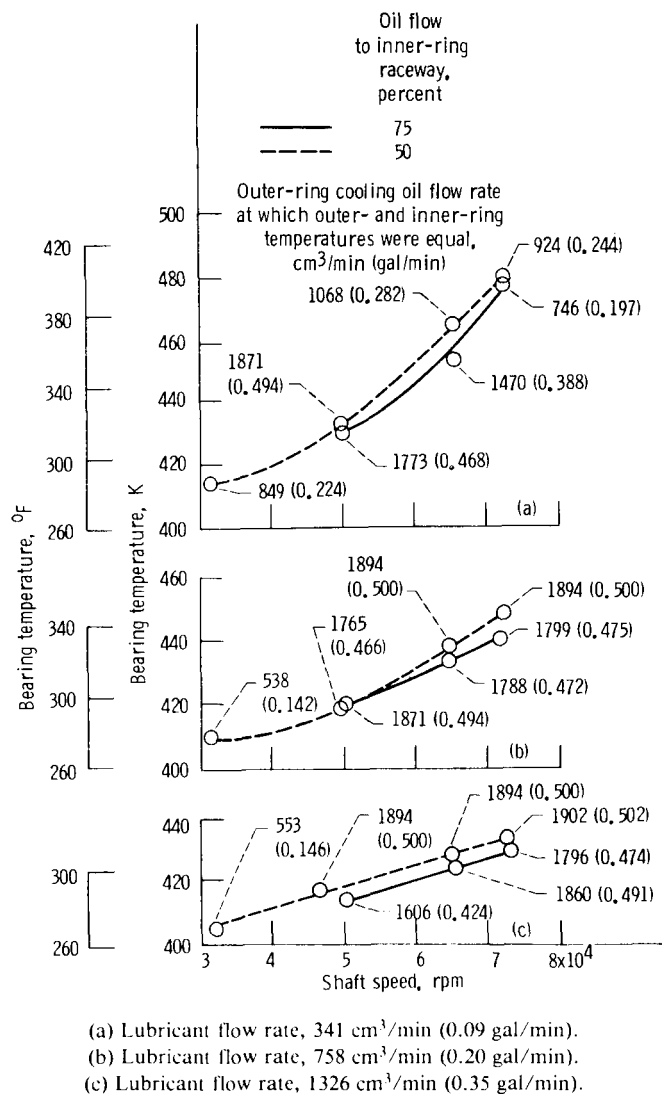


Figure 7.—Effect of shaft speed on test bearing temperature for two oil-flow distribution schemes—outer ring cooled to equal inner-ring temperature. Thrust load, 667 N (150 lb).

curves for the two oil-flow distribution schemes averaged about 5 deg K (9 deg F), except for a point at about 53 000 rpm at 758-cm<sup>3</sup>/min (0.20-gal/min) flow rate (fig. 7(b)), where the two curves intersect.

Cooling the outer ring did not appreciably affect inner-ring temperatures (fig. 8). With 50-percent flow to the inner-ring raceway (fig. 8(a)), the bearing inner-ring temperatures with no outer-ring cooling were almost identical to those with outer-ring cooling. A like comparison of temperature can be made at 1326 cm<sup>3</sup>/min (0.35 gal/min) (fig. 8(b)), as the indicated temperatures show. The advantage of cooling the outer ring can be illustrated by a study of figure 8(a). With 50-percent flow to the inner-ring raceway and no outer-ring cooling, the temperature of the outer ring was 483 K (409 °F) at a speed of 72 000 rpm. When outer-ring cooling was used, this temperature decreased to 448 K (346 °F), a drop in

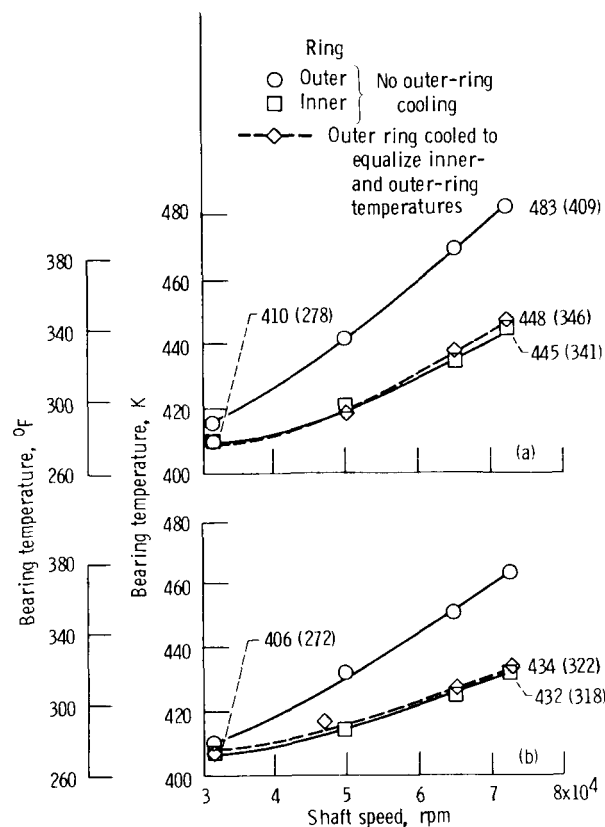


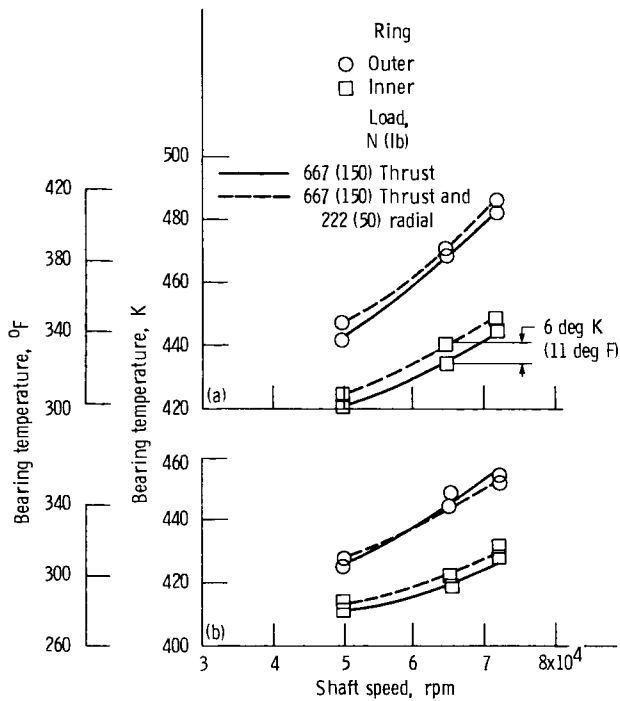
Figure 8.—Effect of outer-ring cooling on bearing temperature at various speeds. 50 Percent of total oil flow supplied to inner-ring raceway; oil-inlet temperature, 394 K (250 °F); thrust load, 667 N (150 lb).

temperature of 35 deg K (63 deg F), or a 7-percent decrease in absolute bearing temperature.

### Effect on Bearing Temperature of Shaft Speed at Two Load Conditions

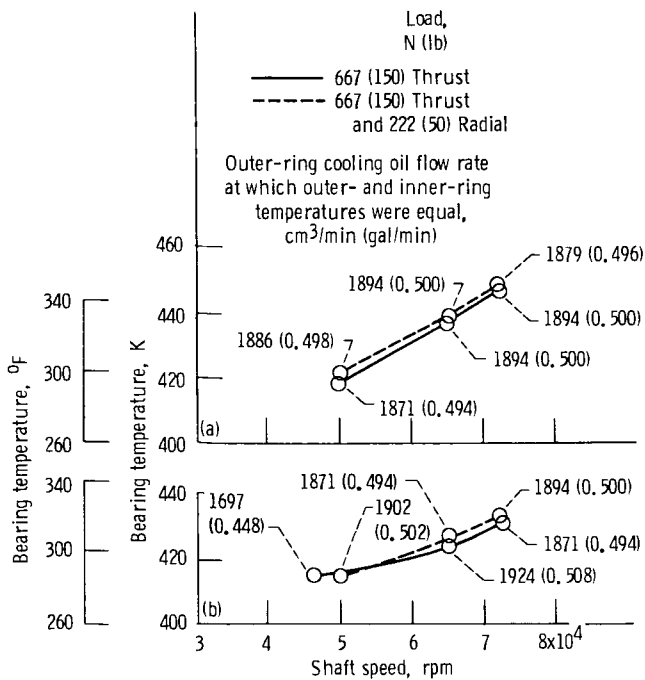
The effect of speed on bearing temperature was determined without and with outer-ring cooling (figs. 9 and 10, respectively). In these tests, 50 percent of the total oil supplied to the inner ring flowed radially through the bearing to the inner-ring raceway. The bearing loads were either a 667-N (150-lb) thrust load or a combined load of 667 N (150 lb) thrust and 222 N (50 lb) radial. Tests were run at total oil flow rates of 758 and 1894 cm<sup>3</sup>/min (0.20 and 0.50 gal/min). Test bearing temperatures increased with increasing shaft speed and decreased with increasing lubricant flow rate for both load conditions with or without outer-ring cooling. Without outer-ring cooling the inner-ring temperature was lower than the outer-ring temperature for all speeds and flow rates tested (fig. 9). Generally, the bearing with a combined load registered slightly higher temperature than the bearing with a thrust load only. The greatest





(a) Lubricant flow rate, 758 cm<sup>3</sup>/min (0.20 gal/min).  
 (b) Lubricant flow rate, 1894 cm<sup>3</sup>/min (0.50 gal/min).

Figure 9.—Effect of shaft speed on test bearing temperature for two load conditions—no outer-ring cooling. Oil-inlet temperature, 394 K (250 °F); 50 percent of total oil flow supplied to inner-ring raceway.



(a) Lubricant flow rate, 758 cm<sup>3</sup>/min (0.20 gal/min).  
 (b) Lubricant flow rate, 1894 cm<sup>3</sup>/min (0.50 gal/min).

Figure 10.—Effect of shaft speed on test bearing temperature for two load conditions—outer ring cooled to equal inner-ring temperature. Oil-inlet temperature, 394 K (250 °F); 50 percent of total oil flow supplied to inner-ring raceway.

difference between the inner-ring temperatures due to loading was 6 deg K (11 deg F) (fig. 9(a)).

Comparing figures 9 and 10 shows that adding outer-ring cooling to thermally balance the inner and outer rings affected the outer-ring temperature much more than the inner-ring temperature, as was previously indicated. The cooling oil flow rates required to produce a thermally balanced bearing ranged from 1697 to 1924 cm<sup>3</sup>/min (0.45 to 0.51 gal/min) for the range of test conditions given in figure 10. Comparing bearing temperatures for the thrust load only with those of the combined load tests shows no appreciable difference over the entire speed range with outer-ring cooling.

### Approaches to Determining Bearing Power Loss

Bearing power loss was determined from both outer-ring bearing torque measurements and calculations of heat rejected to the lubricant. Although the strain gage and the related torque mechanism were calibrated and checked before each test run, hysteresis losses in the knife-edges, restraints due to lubrication and thermocouple lines, and rig vibrations could cause some inaccuracies in the torque readings.

Heat rejection to the lubricant accounts for a major portion of the bearing power loss but does not include all the heat lost by conduction, radiation, and convection. Because of this fact and possible inaccuracies in oil-inlet and oil-outlet temperature measurements, which depend on how close the thermocouples are to the bearing, heat rejected to the oil is only an indication of bearing power loss.

So that heat rejection could be measured, oil-inlet and oil-outlet temperatures were obtained for all flow conditions. The heat absorbed by the lubricant was obtained from the standard heat-transfer equation

$$Q_T = MC_p(t_{out} - t_{in})$$

where

$Q_T$  total heat-transfer rate to lubricant, J/min (Btu/min)

$M$  mass flow rate, kg/min (lb/min)

$C_p$  specific heat, J/kg K (Btu/lb °F)

$t_{out}$  oil-outlet temperature, K (°F)

$t_{in}$  oil-inlet temperature, K (°F)

**Bearing power loss for two oil-flow distribution schemes.**—The power loss in the bearing was determined from torque readings taken with a strain gage and from heat rejection to the lubricant (figs. 11 and 12) for two oil-flow distribution schemes. Bearing power loss increased with increasing speed and lubricant flow rate

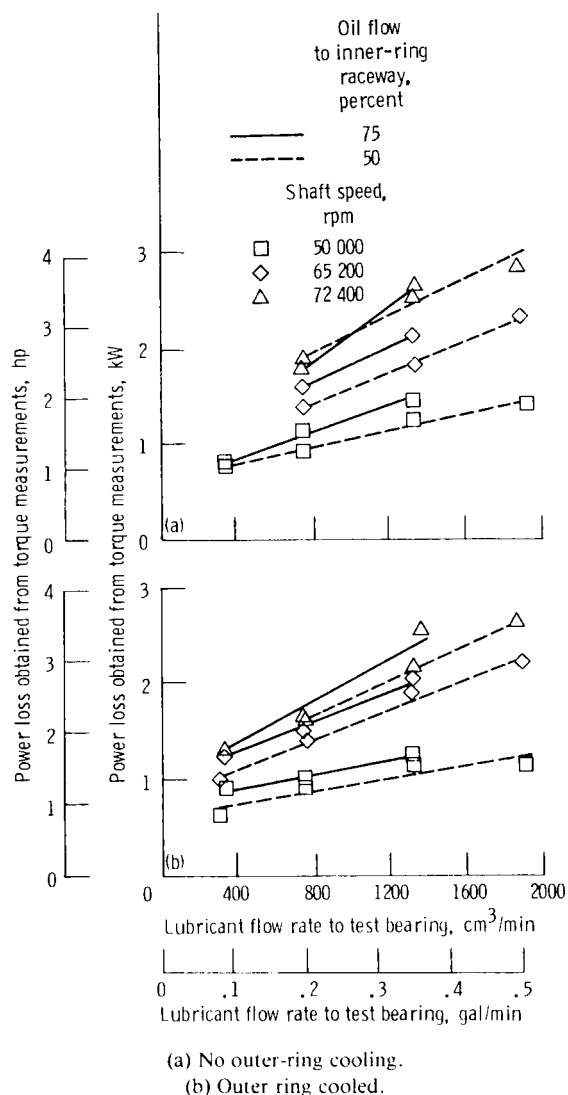


Figure 11.—Effect of lubricant flow rate to inner ring on bearing power loss as obtained from torque measurements, for two oil-flow distribution schemes. Thrust load, 667 N (150 lb).

for both the 50- and 75-percent oil-flow distribution schemes.

Most of the data in figures 11 and 12 show no significant difference in power loss between 50- and 75-percent radial flow to the inner-ring raceway. It is possible that neither power loss measuring technique was sensitive enough to accurately measure the difference with such small changes (shallow-slope curves) in flow rate to the raceway. The maximum power loss obtained for the two oil-flow distribution schemes was 3.0 kW (4.0 hp) as determined from the heat rejected to the oil at 72 400 rpm at a lubricant flow rate of 1840 cm<sup>3</sup>/min (0.49 gal/min) with 50 percent of the flow supplied to the inner-ring raceway and outer-ring cooling (fig. 12(b)). The values of power loss determined from heat rejection

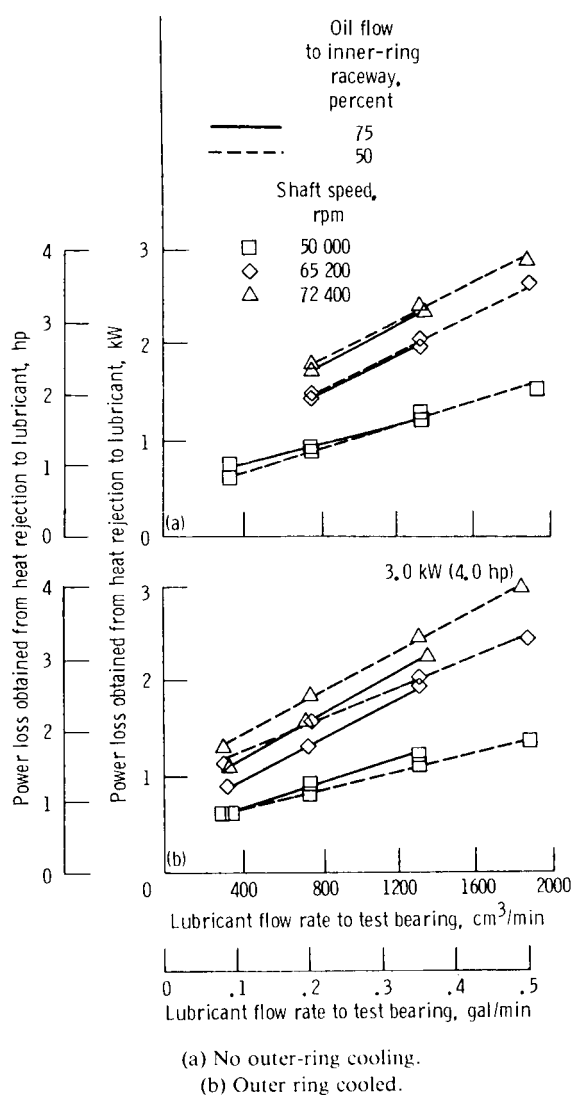


Figure 12.—Effect of lubricant flow rate to inner ring on bearing power loss as obtained from heat rejection to lubricant, for two oil-flow distribution schemes. Thrust load, 667 N (150 lb).

to the lubricant were comparable in magnitude to those obtained from torque measurements.

**Bearing power loss at two load conditions.**—The bearing power loss was determined from torque readings and from heat-transfer calculations (figs. 13 and 14) for two load conditions. In these tests 50 percent of the lubricant was supplied to the inner-ring raceway. Slightly higher values of power loss resulted from a combined radial and thrust load than from a thrust load only. The combined-load and thrust-load-only power loss data were so similar that a single average curve could have been drawn through the data at each speed to represent both load conditions. The maximum power loss recorded under the two load conditions was 3.1 kW (4.2 hp) (fig. 13(a)).

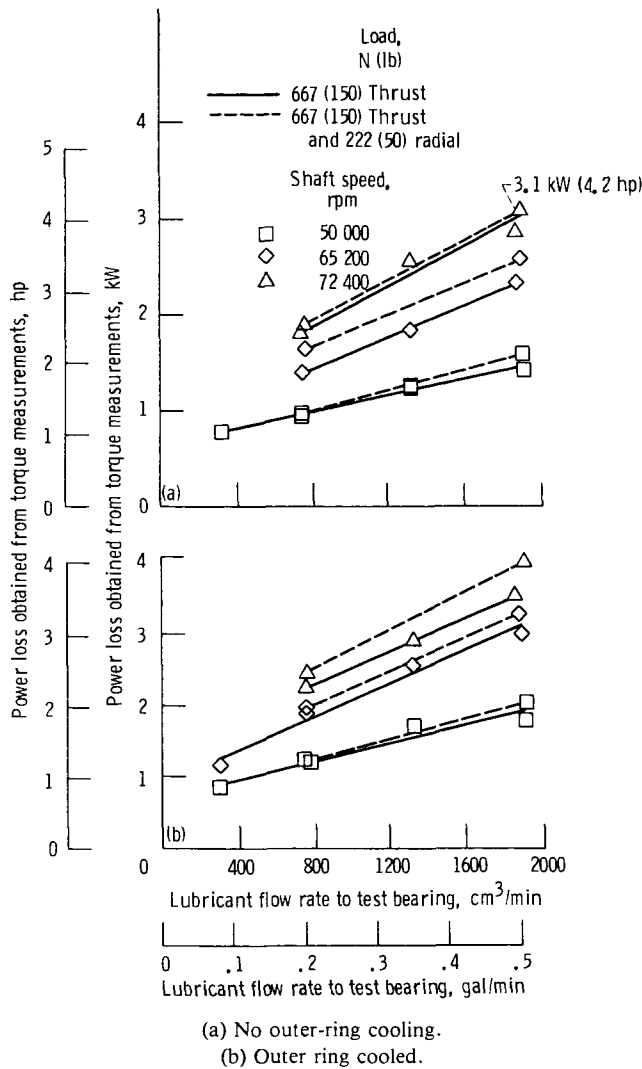


Figure 13.—Effect of lubricant flow rate to inner ring on bearing power loss as obtained from torque measurements, for two load conditions. 50 Percent of total oil flow supplied to inner-ring raceway.

An important conclusion to be drawn from the tests described herein is that outer-ring cooling can significantly reduce bearing operating temperature (by as much as 7 percent, fig. 8), without any appreciable change in bearing power loss.

### Cage Slip

**Effect on cage slip of shaft speed.**—So that percent cage slip could be determined, the epicyclic cage speed  $C_{epi}$  at the various test speeds and load conditions was obtained from SHABERTH (ref. 6), a computer program that took into account centrifugal force effects on contact angle. Elastic contact forces were considered in a race-controlled solution. The epicyclic cage speed was combined with the measured experimental cage speed  $C_{exp}$  to obtain percent cage slip as follows:

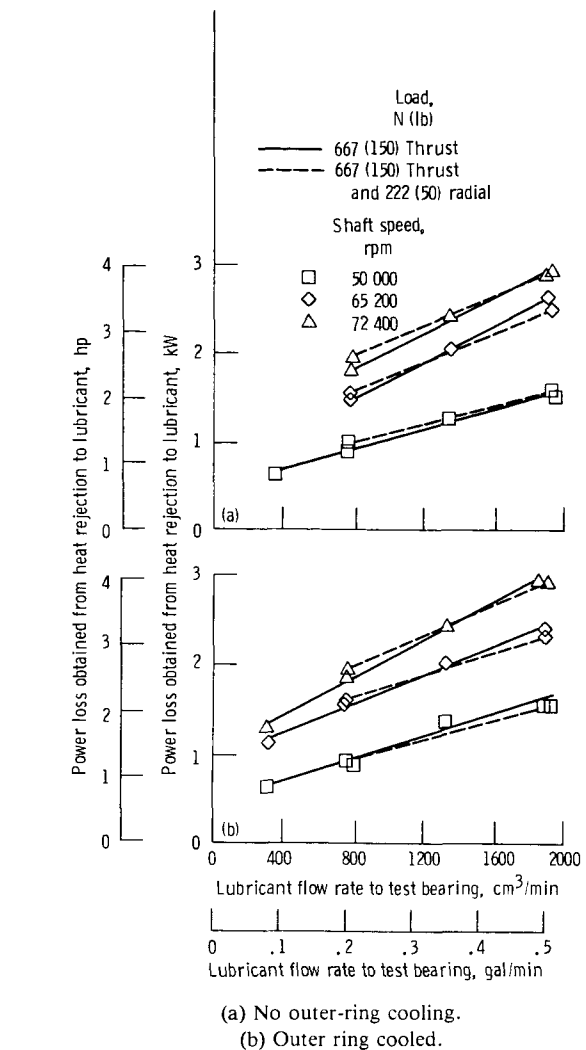
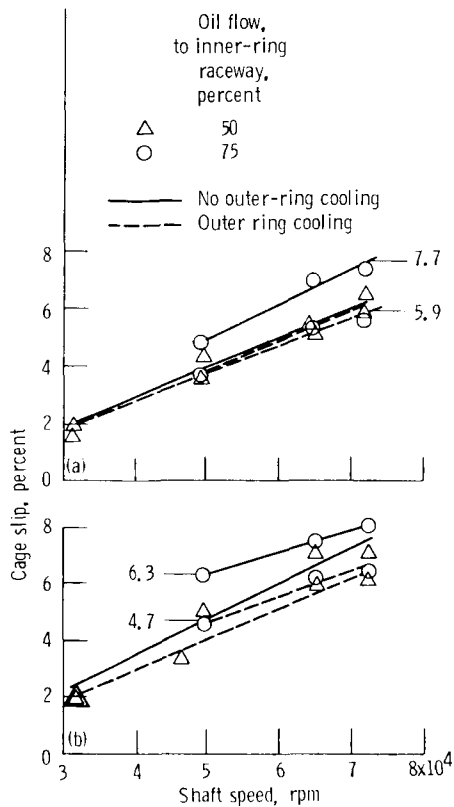


Figure 14.—Effect of lubricant flow rate to inner ring on bearing power loss as obtained from heat rejection to lubricant, for two load conditions. 50 Percent of total oil flow supplied to the inner-ring raceway.

$$\text{Percent cage slip} = \left(1 - \frac{C_{exp}}{C_{epi}}\right)(100)$$

The effect of shaft speed on percent cage slip was determined at lubricant flow rates of 760 and 1330 cm<sup>3</sup>/min (0.20 and 0.35 gal/min) with a thrust load of 667 N (150 lb) (fig. 15). Data obtained with 50- and 75-percent lubricant flow to the inner-ring raceway are compared. The effect of outer-ring cooling on percent cage slip is also shown. For both lubrication flows the percent cage slip increased with increasing shaft speed and lubricant flow.

The bearing with 75-percent lubricant flow to the inner-ring raceway registered the higher percent cage slip for all cases. The maximum difference in cage slip



(a) Lubricant flow rate, 760 cm<sup>3</sup>/min (0.20 gal/min).  
 (b) Lubricant flow rate, 1330 cm<sup>3</sup>/min (0.35 gal/min).

Figure 15.—Effect of shaft speed on cage slip for two oil-flow distribution schemes, with and without outer-ring cooling. Thrust load, 667 N (150 lb).

between the 50- and 75-percent oil-flow distribution schemes was 1.6 (6.3–4.7) percent and occurred at a shaft speed of 50 000 rpm and a lubricant flow rate of 1330 cm<sup>3</sup>/min (0.35 gal/min) without outer-ring cooling (fig. 15(b)). Cooling the outer ring reduced the percent cage slip for both oil-flow distribution schemes (fig. 15). Cage slip decreased by as much as 1.8 (7.7–5.9) percent with outer-ring cooling at a shaft speed of 72 000 rpm and a lubricant flow rate of 760 cm<sup>3</sup>/min (0.20 gal/min) with 75-percent radial flow to the inner-ring raceway (fig. 15(a)).

Increasing lubricant flow rate increased the percent cage slip, as might have been expected, because of the additional drag on the balls and the cage. The observed increase in cage slip with increasing shaft speed might have been expected because centrifugal forces decrease the ball load and the traction at the inner-ring raceway contact.

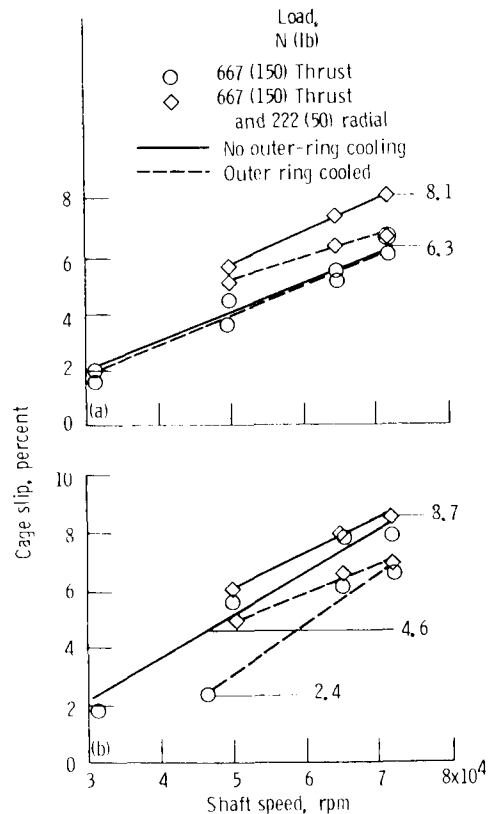
**Effect on cage slip of adding radial load to thrust load.**—A 222-N (50-lb) radial load was added to the 667-N (150-lb) thrust load to determine the effect on percent cage slip over the test speed range. Lubricant flow rates were 760 and 1900 cm<sup>3</sup>/min (0.20 and 0.50 gal/min) with 50 percent of the total oil flow to the

bearing directed radially to the inner-ring raceway. The bearing was run with and without outer-ring cooling. The added radial load increased cage slip by a maximum of 1.8 (8.1–6.3) percent at a nominal speed of 72 000 rpm (fig. 16(a)). Apparently the addition of a radial load relieves the traction force between the balls and bearing rings sufficiently to increase the amount of slip in the bearing.

When the outer ring was cooled, percent cage slip decreased by a maximum of 2.2 (4.6–2.4) percent at a nominal speed of 46 500 rpm (fig. 16(b)). The maximum cage slip recorded for all tests was 8.7 percent at 72 000 rpm (fig. 16(b)).

The plots of percent cage slip versus shaft speed shown in figures 15 and 16 had been expected to go through zero origin. This was shown in reference 7 where the data curves of a similar 35-mm-bore ball bearing were plotted at lower speeds and then extrapolated through zero origin.

Visual examination of the bearing after running showed no damage to the raceways, balls, or cage. Therefore the cage slip that did occur was not of sufficient magnitude to affect bearing operation adversely.



(a) Lubricant flow rate, 760 cm<sup>3</sup>/min (0.20 gal/min).  
 (b) Lubricant flow rate, 1900 cm<sup>3</sup>/min (0.50 gal/min).

Figure 16.—Effect of load and outer-ring cooling on percent cage slip at various shaft speeds. 50 Percent of total oil flow supplied to inner-ring raceway.

## Summary of Results

Parametric tests were conducted with a 35-mm-bore ball, split-inner-ring bearing with a double-inner-land-guided cage. Test conditions included loads of 667-N (150-lb) thrust or a combined load of 222 N (50 lb) radial and 667 N (150 lb) thrust, shaft speeds from 32 000 to 72 000 rpm, lubricant flow rates from 341 to 1894 cm<sup>3</sup>/min (0.09 to 0.50 gal/min), and outer-ring cooling rates from 0 to 1932 cm<sup>3</sup>/min (0.51 gal/min). Provisions were made for through-the-inner-ring lubrication. In some tests 50 percent of the oil supplied to the inner ring flowed radially through the bearing to the inner-ring raceway to lubricate the bearing, and 25 percent flowed axially through grooves with no radial holes to cool the bearing inner ring. The remaining 25 percent flowed radially through the bearing for cage-land lubrication. In other tests this flow distribution was 75, 0, and 25 percent, respectively.

The following major results were obtained:

(1) Bearings with two different oil-flow distribution schemes to lubricate and cool the bearing were successfully operated to 2.5 million DN.

(2) The portion of the total oil flow (25 percent) used for cooling only the inner ring was essentially ineffective. However, cooling the outer ring to produce a thermally balanced bearing (with equal inner- and outer-ring temperatures) decreased outer-ring absolute temperature by 7 percent at 2.5 million DN.

(3) The maximum power loss observed herein was 3.1 kW (4.2 hp) with a combined load, a shaft speed of 72 400 rpm, and a lubricant flow rate of 1856 cm<sup>3</sup>/min (0.49 gal/min) with 50-percent flow to the inner-ring raceway and without outer-ring cooling.

(4) Percent cage slip increased with increasing shaft speed and lubricant flow rate and was greater at 75- than at 50-percent lubricant flow to the inner ring.

(5) Adding a 222-N (50-lb) radial load to the 667-N (150-lb) thrust load increased the percent cage slip by a maximum of 1.8 percent. The maximum cage slip recorded for all tests was 8.7 percent. The largest drop in cage slip due to outer-ring cooling was 2.2 percent.

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